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Modelling and transient response study of hydraulic servo actuator

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Modelling and transient response study of hydraulic servo actuator

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Abstract. This paper presents a mathematical model of electro hydraulic servo system (EHS) using two-stage electrohydraulic servovalve (EHSV), by the aim of Matlab-simulink-simscape multibody to predict the flow characteristics (pressure/flow rate) with changing the parametric configuration of (EHSV), these parameters could be used to estimate the error at different operating conditions. It includes establishing mathematical model, controller model and validating the performance of the (EHSV) by experimental result of published work using the data of servovalve labelled B.31.210.12.1000.U2V (mechanical feedback) manufactured by PPT – Trstenik. The effect of changing servovalve orifice diameter (0.25, 0.30, and 0.35) mm, show that increasing the orifice diameter of the servovalve leads to increase the transient time but decrease the system overshoot. The pressure behaviour is plotted and shows that increasing the orifice diameter leads to increasing the pressure inside the system as to reach the relieve pressure.

1. Introduction

The electro hydraulic servo systems (EHS) are the best solution for many applications due to its power supply for large force, its accuracy and its response. There are many applications of (EHS) such as remote lifting, materials handling, machine tool drives, flight control, rolling mills, aircraft simulator and etc. [1]. Servo valves are used in controlling the hydraulic circuits where the force or position on the hydraulic cylinder is measured and feed back to a controller that varies the signal sent to the servo valve which allows very precise position control.

Modelling of two-stage electrohydraulic servo valve with spool position feedback was created in [2], and the model were verified experimentally using (B.31.210.12.1000.U2V) Moog servovalve. This model can predict the performance of the servo valve in a wide range of expected working conditions.
A development of an improved servovalve model with both leakage and orifice flows were established in [3], it is verified experimentally.

Derivation, simulation, and implementation of a nonlinear tracking control law for a hydraulic servo system was made by [4], it provides a controller that achieve a position tracking which is simulated and experimentally tested to test the limits of its performance and the realistic effects of friction.
The flow through spool orifices in different working regimes were analytically described in [5] which calculate the internal leakage through the spool orifice.
Dynamic modelling of an electro hydraulic servo actuator with constant load was established with a controller model in [6], which assigned a control parameter on the experimental model by using fuzzy PID control controller.

Motion control of an aircraft electro-hydraulic servo actuator was obtained in [7], it provided a mathematical model of an aircraft integrated electro-hydraulic servo-actuator and designed a classical PID controller by Zeigler-Nichols tunning method.
An online controller with least-mean-square algorithm is controlled to the force closed-loop system of the electro hydraulic servo system controlled by offline designed controller in [8].
Mathematical model is developed in [9], with the characteristics of pump, hydraulic cylinder, dynamic characteristic of the valve and position feedback. Adaptive Neuro Fuzzy is used to control servo valve with position feedback (-10 to 10 V) made by Rexroth Bosch to control a hydraulic actuator in [10].

2. Model description

The system **illustrated in figure (1), consists of hydraulic power unit (hydraulic pump), which delivers hydraulic to the control unit (hydraulic servo valve) which is adapted by the controller (PID), to deliver the required hydraulic to the hydraulic actuator, which track the required trajectory.**

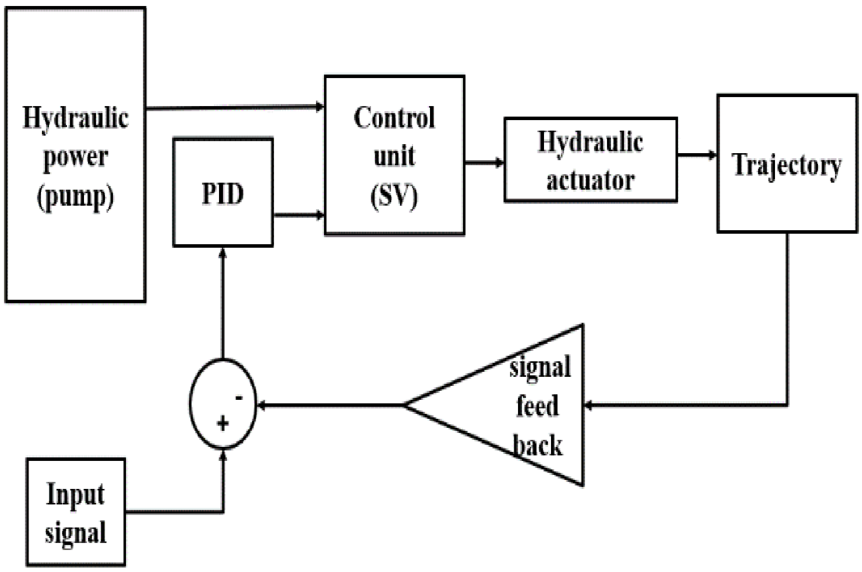

Figure 1. block diagram for the overall system

Figure (2) focus on the hydraulic circuit of the proposed system which is composed of a source of hydraulic (hydraulic pump) (1), connected with pressure relieve valve (2), that delivers fluid to the electro hydraulic servo valve (EHSV) (3) via a one way valve (4), (EHSV) controls the output hydraulic that delivers to the hydraulic actuator (5) to allow the hydraulic actuator tracking the required input position by the aim of PID controller (6), that take a feedback from position sensor (LVDT) (7) the figure illustrates the hydraulic circuit and the specification of this circuit is shown in table (1).

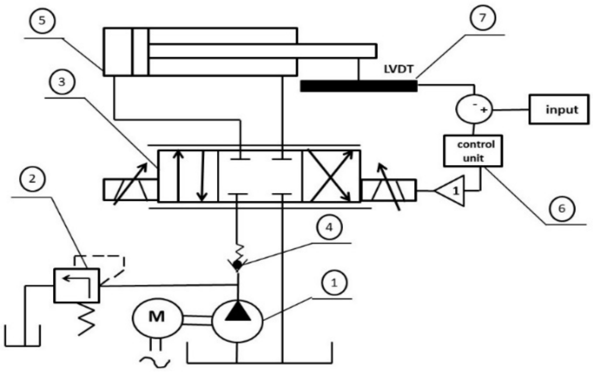

Figure 2. Electro hydraulic servo actuator circuit.

Figure (3-a) illustrates a typical servo valve used in this study where it consists of torque motor (1), that drives flapper (2), which makes a difference in pressure between servovalve nozzles (3), to force the sliding spool (4), to allow pressure port (5), to feed port (6) or (7).
The spool other side allows hydraulic to go back through the return port (8), while the two orifices (9) apply a pressure difference.

Figure (3-b) focuses on the servo valve torque motor which contains:

1 polar piece
2 magnet
3 flexible tube
4 flapper
5 control coils
6 Armature

While figure (3-c) illustrates the flapper-orifice area of the servo valve to focuses on the important of this area on the hydraulic behaviour, it also illustrates the valve input flow rate, flow rates before valve orifices (Q_1 and Q_2) and flow rates after valve orifices (Q_3 and Q_4).

Table 1. Parameters table.

Hydraulic element	symb	value	Hydraulic element	symb	value
Hydraulic cylinder	Ap (A)	1.2e-3 (m^2)	Pump	Q	12 (L/min)
	Ap (B)	1.2e-3 (m^2)		n	1800 (rpm)
	L	0.5 (m)		V_g	7.07e-7 (m3/rad)
	V_0	1e-5 (m^3)		η_v	92 %
Servovalve	As	16.76 (mm^2)	System	B	1.5e9 (Pa)
	D_n	0.28 (mm)		P_s	210 (bar)
	m	3.1 (Kg)		P_T	0 (bar)
	D_o	0.18 (mm)		ρ	871 (kg/m3)

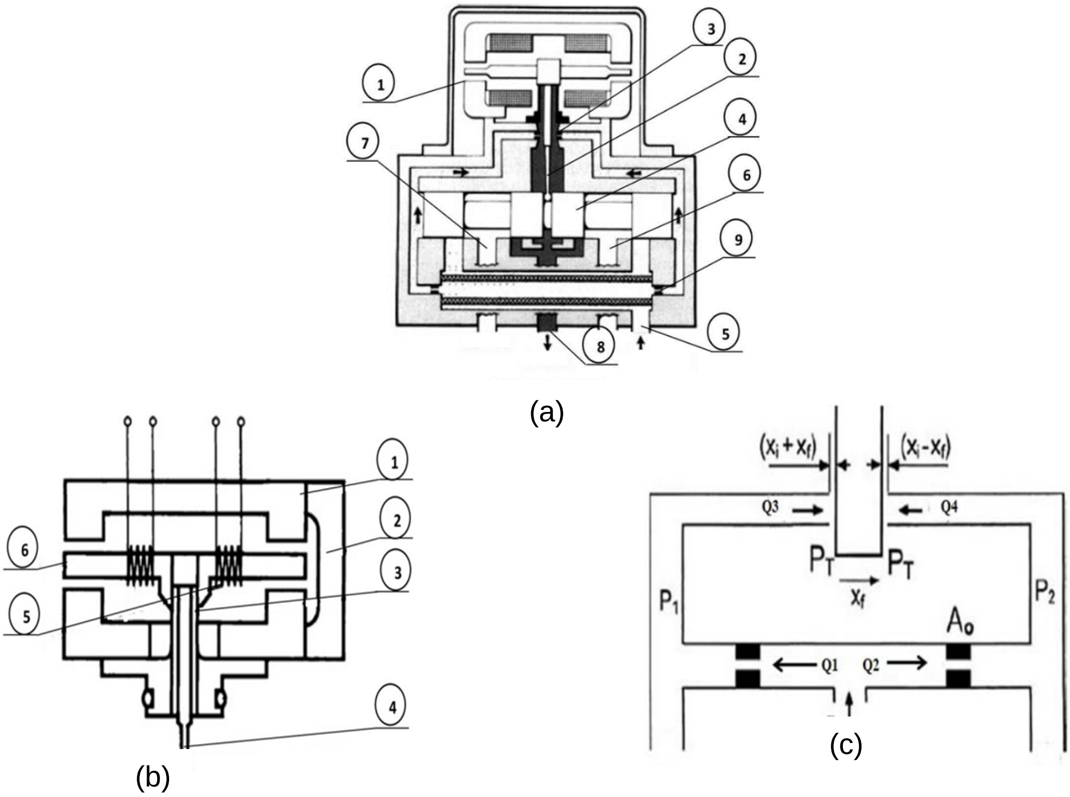

Figure. 3: Typical two stage electro hydraulic servo valve. (a) Valve cross section. (b) Valve torque motor. (c) Orifice and flapper section.

3. Mathematical model

The servo valve mathematical model is derived in 2009, by G. Rabie, [11].
The effective torque delivered by the electromagnetic motor is given by the equations (1) and (2).

$$\mathrm{T} = \mathrm{K}_i * \mathrm{i} + \mathrm{K}_\theta * \theta \tag{1}$$

Where

$$\mathrm{K}_i = \frac{N\, \lambda_p \mu_0\, \mathrm{AL}}{2{X_0}^2}$$

$$\mathrm{K}_\theta = \frac{{\lambda_p}^{\mathbf{2}}\, \mu_0\, \mathrm{AL}^2}{4{X_0}^3}$$

i input current. N. number of turns of the coil, λp... magneto-motive force of the permanent magnet, μ0... permeability of free space, A... area of air gap.

L.. armature length, X_0.. length of the air gap in the neutral position of the armature.
The armature equation of motion is

$$\mathrm{T} = J\frac{d^2\theta}{dt^2} + \mathrm{f}_v\frac{d\theta}{dt} + \mathrm{K}_t\theta + \mathrm{T}_l + \mathrm{T}_p + \mathrm{T}_F \tag{2}$$

The flow rate affecting on the flapper and make a pressure difference between both sides of the spool is obtained by equations (3), (4) and (5).

$$\mathrm{Q}_1 = \mathrm{C}_d\mathrm{A}_0\sqrt{\frac{2}{\rho}}(\mathrm{P}_s - \mathrm{P}_1) \qquad \mathrm{Q}_2 = \mathrm{C}_d\mathrm{A}_0\sqrt{\frac{2}{\rho}}(\mathrm{P}_s - \mathrm{P}_2) \tag{3}$$

$$Q_3 = C_d \pi\, d_f \left(X_i + X_f\right)\sqrt{\frac{2}{\rho}}(P_1 - P_3) \tag{4}$$

$$Q_4 = C_d \pi\, d_f \left(X_i + X_f\right)\sqrt{\frac{2}{\rho}}(P_2 - P_3) \tag{5}$$

The flapper chamber continuity equations had shown in equations (6), (7) and (8).

$$Q_1 - Q_3 + A_s \frac{dx}{dt} = \frac{V_0 - A_s\, x}{B} \frac{dP_1}{dt} \tag{6}$$

$$Q_2 - Q_4 - A_s \frac{dx}{dt} = \frac{V_0 + A_s\, x}{B} \frac{dP_2}{dt} \tag{7}$$

$$Q_3 + Q_4 = \frac{V_3}{B} \frac{dP_3}{dt} \tag{8}$$

The spool equation of motion is shown in (9)

$$A_s(P_2 - P_1) = m_s \frac{d^2x}{dt^2} + f_s \frac{dx}{dt} + F_j + F_s \tag{9}$$

Continuity equations through the hydraulic actuator:

$$Q_{AR} - a\frac{dy}{dt} - Q_i - Q_e - \frac{(V_R + ay)}{B} \times \frac{dp_R}{dt} = 0 \tag{10}$$

$$A\frac{dy}{dt} + Q_i - Q_{psB} - \frac{(V_{ps} - Ay)}{B} \times \frac{dp_R}{dt} = 0 \tag{11}$$

Actuator Equation of motion:

$$a.p_R - A.p_{ps} = m_c \frac{d^2y}{dt^2} + f_c \frac{dy}{dt} + F \tag{12}$$

These equations represent the mathematical model which are presented by the aim of Matlab-simulink, then using System Identification Toolbox and hence the transfer function is defined as

$$\frac{63.12}{S^2 + 16.44S + 63.09} \tag{13}$$

Then by trial and error with PID controller toolbox system tunned, the three coefficients (KP,KI and KD) are adopted to KP (20) KI (1) KD (0.01).

4. validation

The performance of (EHSV) is validated by experimental result of published work in [2] as seen in figure (4), with the same data in table (1) by applying ±10% of the rated current signal, with square periodic signal (frequency was 10 Hz).

The result shows that the simulation result have a good agreement with the experimental one.

The validated simulation of (EHSV) has been used to estimate the proposed servo actuator performance in this study.

5. Result and discussion

The controller is responsible for reducing error during running condition as it adopts the input signal to compensate the required position and also adopts the transient response of the actuator.

In this study attempts to predict the response of the actuator with varying the geometric configurations of the control unit (EHSV) is done by changing valve orifice diameter to simulate the effect of impurities such as burrs, metal particles, and particles produced during operation on servo valve operating condition, these types of impurities are considered in this study to discuss its effect on servovalve response after long time of operation and hence predict the servovalve life time.

Figure (5) shows the position tracking of three servo valves with three different orifice diameters (0.25, 0.30and 0.35) mm.

As it shown when using orifice diameter 0.25 mm it effects on the overshoot of the actuator to reach 0.35 m and also reach the steady state condition after 0.6 second.

By using orifice diameter 0.30 mm the actuator overshoot is reduced to 0.31 m and the settling time is increased to 0.65 second. By using orifice diameter 0.35 mm the actuator overshoot can be neglected and the settling time is increased to 0.8 second. This increasing in the settling time is due to increasing the pressure difference caused by the orifice to reach the relief valve pressure (200 bar) as shown in figure (8). This behaviour "by using step input" is obvious appeared and also by using pulse input, same overshoot and same delay is done as shown in figure (6).

Figure (7) and (8) show the pressure variation through the servovalve orifices (left and right side) while acting with input signal 0.3 m and it explains that by increasing the orifice diameter to 0.30 and 0.35 mm the pressure increases in the spool left side to the relief pressure which is not suitable to compensate the required force to push the spool of (EHSV) to deliver fluid to the actuator, while by using orifice diameter 0.25 the system is stable.

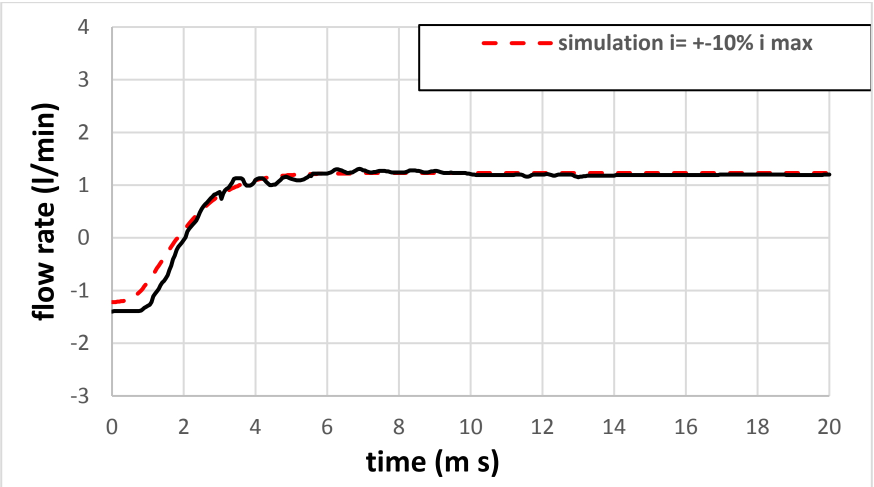

Figure. 4: Experimental and simulation servovalve flowrate behavior with input signal ±10% of rated current signal

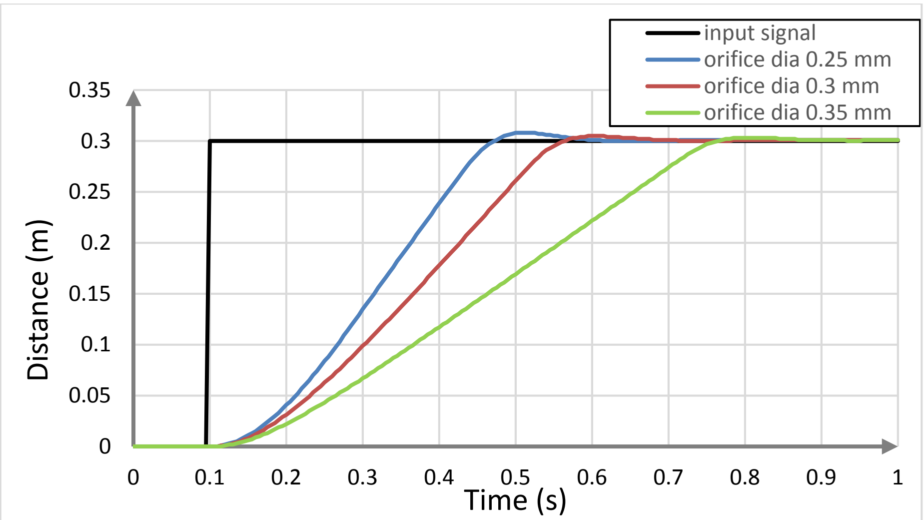

Figure 5: position tracking by step input 0.3 m with varying the orifice diameter (0.25, 0.30 and 0.35) mm.

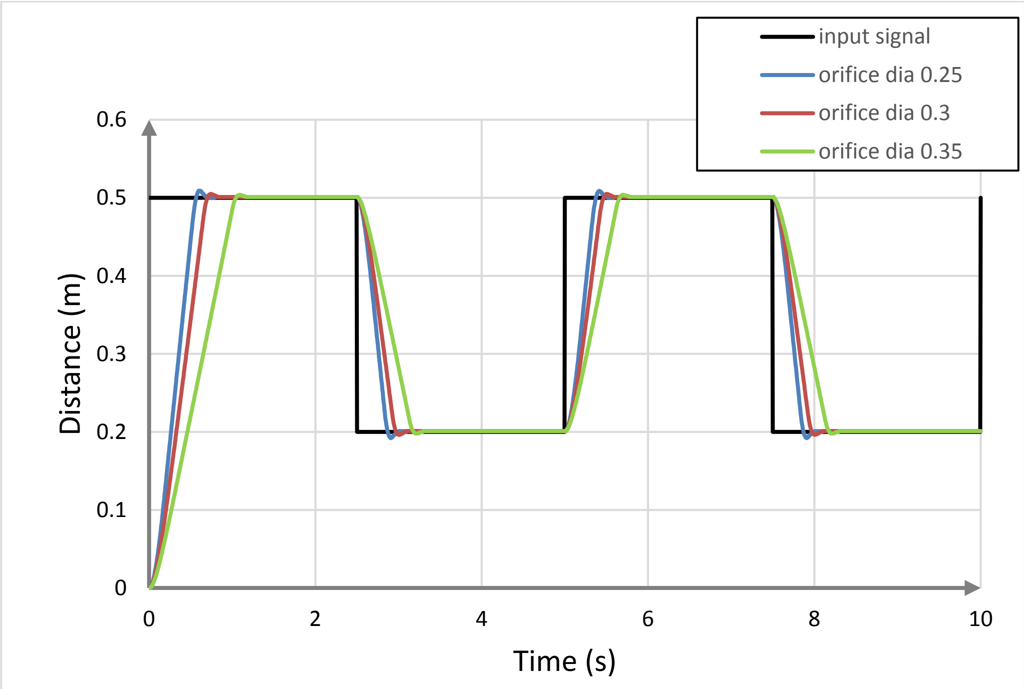

Figure 6: position tracking by square pulse input with varying the orifice diameter (0.25, 0.30 and 0.35) mm.

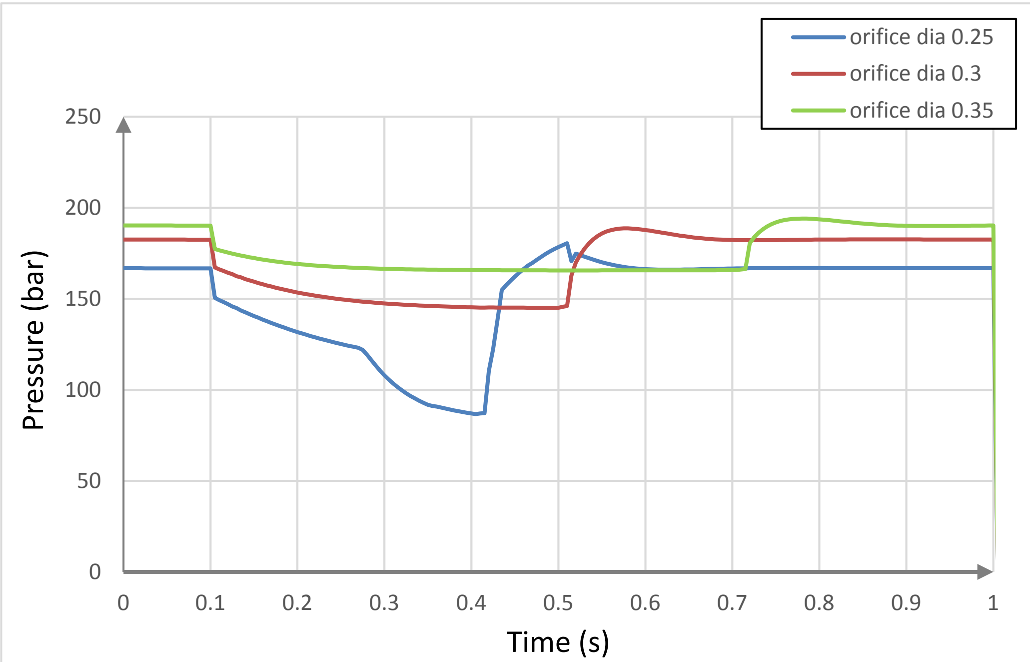

Figure 7: pressure through servo valve, input displacement (0.3 m) with varying the orifice diameter (0.25, 0.3, and 0.35) mm. left side.

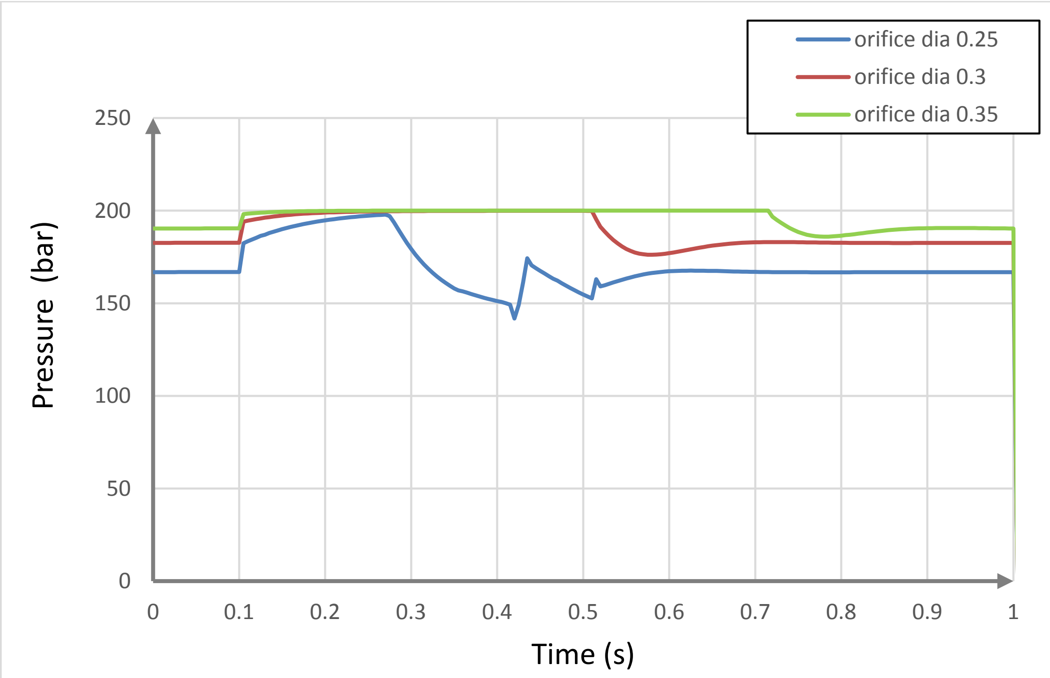

Figure 8: pressure through servo valve, input displacement (0.3 m) with varying the orifice diameter (0.25, 0.3, and 0.35) mm. right side.

6. Conclusion

The electro hydraulic servo valve is a very sensitive hydraulic control unit as it is affected by the very low sized impurities due to its tiny orifices' configuration.

It is deduced that the diameter of servo valve orifice effects on the transient time response of the system as if its orifice diameter is exposed to any small change it effects on the transient time and also effects on the system overshoot. That is due to the variation in the pressure behaviour due to the change in the orifice area.

Nomenclature

Q	flow rate
C_d	discharge coefficient
A_0	orifice area
d_f	flapper nozzle diameter.
P_s	feeding pressure.
P_T	returns pressure.
X_f	flapper displacement.
X_i	initial flapper nozzle opening.
ρ	oil density.
A_s	spool cross-sectional area.
B	bulk modulus.
V_o	initial volume of oil in the spool side.
V_3	volume of the flapper valve return chamber.
L	actuator stroke.
V_g	geometric volume.
η_v	volumetric efficiency.
D_n	nozzle diameter.
m	spool mass.
D_o	orifice diameter.
J	moment of inertia

T_l torque due to flapper displacement limiter
f_v damping coefficient
K_t stiffness of flexure tube
T_F feedback torque
T_p torque due to the pressure forces.

References

[1] M. A. S. Aboelela, M. E. S. M. Essa, and M. A. M. Hassan, 2018 “Modeling and identification of hydraulic servo systems,” *Int. J. Model. Simul*., vol. **38**, no. 3, pp. 139–49.
[2] Gordić, Dušan, Milun Babić, and Nebojša Jovičić. 2004, "Modelling of spool position feedback servovalves." *International Journal of Fluid Power* **5.1** pp 37-51.
[3] Eryilmaz, Bora, and Bruce H. Wilson. 2000, "Combining leakage and orifice flows in a hydraulic servovalve model." *J. Dyn. Sys., Meas., Control* **122** pp: 576-79.
[4] Sohl, Garett A., and James E. Bobrow. 1999, "Experiments and simulations on the nonlinear control of a hydraulic servosystem." *IEEE transactions on control systems technology* **7.2** pp: 238-47.
[5] Gordić, D., et al. 2011, "Spool valve leakage behaviour." *Archives of Civil and Mechanical Engineering* **11.4** pp: 859-66.
[6] Tran, Ngoc Hai, et al. 2018, "Modeling of Position Control for Hydraulic Cylinder Using Servo Valve." *Asian Conference on Intelligent Information and Database Systems*. *Springer, Cham*.
[7] Fadel, M. Z., M. G. Rabie, and A. M. Youssef. 2019, "Motion control of an aircraft electro-hydraulic servo actuator." *IOP Conference Series: Materials Science and Engineering*. Vol. **610**. No. 1. IOP Publishing.
[8] G. Shen, Z. Zhu, J. Zhao, W. Zhu, Y. Tang, and X. Li, 2017 “Real-time tracking control of electro-hydraulic force servo systems using offline feedback control and adaptive control,” *ISA Trans.*, vol. **67**, pp. 356–370.
[9] V. I. Forental, M. V. Forental, and F. M. Nazarov, “Investigation of dynamic characteristics of the hydraulic drive with proportional control,” Procedia Eng., vol. **129**, pp. 695–701, 2015.
[10] M. E. S. M. Essa, M. A. S. Aboelela, M. A. Moustafa Hassan, and S. M. Abdrabbo, 2019 “Control of hardware implementation of hydraulic servo application based on adaptive neuro fuzzy inference system,” ICENCO 2018 - *14th Int. Comput. Eng. Conf.* Secur. Smart Soc., pp. 168–73.
[11] M. G. Rabie, Fluid power engineering vol. 28: McGraw-Hill New York, NY, USA, 2009